REPORT NO. 66576

EVALUATION OF SIMULATION PROCEDURES FOR HIGH INTENSITY NOISE FIELDS

FINAL REPORT

PREPARED UNDER CONTRACT NO. NAS1-6906 BY

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PREFACE

The Evaluation of Simulation Procedures for High Intensity Noiee Fields has been conducted under Contract No. NAS1-6906 with the NASA Langley Research Center. The program has included an experimental study of the interaction between model structures and sound fields. The results of these studies have been applied to the evaluation of simulation facilities. The project has been conducted under the general guidance of Dr. Robert W. Benson. Mr. S. H. Pearsall and Mr. W. M. Stafford contributed to the experimental program with the technical assistance of Mr. J. Holladay.

ABSTRACT

An experimental program has been conducted to determine those aspects of a sound field which correlate with motion of a structure excited at one of its modes of vibration. It has been demonstrated that the forcing function for a structure is the instantaneous differential pressure existing across the structure. A measurement of pressure gradient or particle velocity as vector quantities therefore correlates with the observed motion. Actual sound fields surrounding typical high intensity noise sources are not described in a manner which allows for a direct evaluation of all simulation facilities. Most facilities require additional measurements in order to determine their adequacy.

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I. INTRODUCTION

The dynamic excitation of mechanical structures results in failure of the structure at stress levels which are low compared to the static strength of the material providing the number of cycles of excitation is large. This particular type of failure of a structure is referred to as fatigue failure. Although direct excitation of a structure is a dominant cause of fatigue failure, acoustically induced vibration is also a cause of fatigue failure especially in jet powered aircraft and rocket powered missiles. Fatigue failure induced by acoustic energy is referred to as sonic fatigue. Since the acoustic energy is large at relatively high frequencies, a large number of cycles can occur in a short period of time, causing fatigue failure at relatively low stress levels.

A second problem associated with acoustically induced vibration is concerned with the malfunction of both electronic and electromechanical assemblies used within aircraft and missiles. In this particular case the induced vibration causes extraneous signals to exist within the equipment resulting in a malfunction of a particular system or assembly. There is a need to evaluate both sub-system assemblies and actual structures used in aircraft and missiles to determine their susceptibility to high intensity noise.

Considerable experience has been derived from the testing of structures and assemblies subjected to vibration caused by direct mechanical excitation. These tests consist of both single frequency and random noise type excitation. For vibration testing, the assembly or structure is subjected to mechanical excitation in each of three orthogonal directions. The testing of an assembly of structure subjected to acoustic excitation involves the placement of the object within an acoustic field which simulates the environment to which the object is exposed under actual conditions.

Several factors have influenced the nature of the facility used for the study of acoustically induced vibration. The actual sound field for various parts of the structure of aircraft and missiles has been measured extensively.[1]

The measurements indicate sound pressure levels in excess of 140 decibels with energy distributed throughout the audio-frequency spectrum. The first requirement for a simulation facility is therefore a sound field having a sound pressure level of 140 decibels with a broad frequency spectrum. In order to avoid an exact duplication of the actual sound field using full scale sources, methods of creating high sound pressures with small amounts of input power have been used. The reverberation room is an example of a facility which has been used extensively for sonic testing where high sound pressure levels are created by the multiple reflection of sound energy from the surfaces of the enclosure.

The use of sound energy to study the mechanical excitation of structures is considerably more complex than the study of human reaction to a sound field. Until the sound intensities reached those levels produced by jet engines, the major interest in high intensity noise was the effect of the noise on hearing. For these studies, the simulation of the pressure aspects of a sound field have been thought to be sufficient. For the study of mechanical excitation by sound waves, it is necessary to give consideration to the exact nature of the sound field since energy must be transferred by the sound field to the structure if damage is to occur.

The present study has been concerned with the measurement of the response of model structures to sound fields where various aspects of the sound field could be varied in a controlled manner. The objective has been to determine that aspect of the sound field which correlates with motion of a structure. A further objective of the program has been to determine those limitations, if any, which are imposed by

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[1] von Gierke, H. E., "Physical Characteristics of Aircraft Noise Sources," The Journal of the Acoustical Society of America, 25, 367 (May 1953).

various types of simulation facilities used for the study of sonic fatigue.

II. THE SOUND FIELD

The description of a sound field requires a definition of particle displacement, velocity and acceleration as well as pressure, all as a function of time. The energy in the sound field propagates in the form of a longitudinal wave for which the particle motion is in the direction of the propagation of energy. The wave may propagate as a straight line (a plane wave) or it may spread out from a point or line source resulting in spherical or cylindrical waves. The waves are subject to scattering and diffraction as well as to the combination of wave systems leading to interference.

When dealing with the transfer of energy from a sound field to a mechanical structure which is immersed in the sound field it is necessary to examine those particular aspects of the field which can result in the transfer of energy. If we consider the simplest case where the disturbance takes place in a single direction, a plane wave, and the wave propagates in the x direction, we may derive those equations which indicate the conveyance of energy by the sound wave. If f denotes the displacement of the particle of the medium from its original position and $\frac{df}{dt}$ the corresponding displacement velocity, then the displacement per unit length is $\frac{df}{dx}$ commonly called the linear strain. If the medium is completely elastic, then the stress or pressure is proportional to strain

$$X = Y \frac{df}{dx}$$

where Y is the elasticity for the medium and X is the force per unit area.

From elementary mechanics the product of stress and displacement velocity is the rate at which energy is transferred by the wave at the point x. The power per unit area is therefore:

$$P = X \frac{df}{dt} \quad \frac{df}{dx}$$

The power therefore varies in both time and space, and is completely specified providing the displacement of the particle is defined for x and t as a function of space. It is further noted that the displacement has both magnitude and direction.

The power conveyed in an acoustic wave is dependent upon two factors with appropriate constants. These factors are the change of displacement with respect to space and the change of displacement with respect to time. Although the pressure may be derived from the above, it is related directly by the elasticity of the medium.

The excitation of a mechanical structure immersed in a sound field implies that the structure converts the acoustic energy into mechanical energy by an interaction of the structure with the sound field. The exact means of energy transfer is dependent upon the mode of vibration of the structure and its relationship to the space and time rates of change of displacement of particles of the medium.

Unfortunately, instrumentation is not available which allows for the direct measurement of the energy conveyed in an acoustic wave nor is it possible to measure directly the energy extracted from a wave by a particular structure. The single instrument which is readily available for accurate measurements of the sound field is the pressure sensitive microphone. This particular transducer allows for a measurement of the temporal variation of the pressure at a given location in the sound field. With the simultaneous use of two pressure sensitive elements, it is possible to measure the relative time histories at two locations in space and deduce from these measurements, the direction of propagation as well as the magnitude of the sound wave.

Before attempting to define those particular aspects of a sound field which should be measured, a useful facility is available which allows for the variation of particular aspects of the sound field in an independent manner. An acoustic tube, commonly used for the measurement of the acoustic impedance of various materials, allows for the propagation of sound energy for which the pressure within the tube may be varied with relation to the particle displacement by the superposition of two waves, one propagating in each direction within the tube. The flow of energy can further be varied by controlling the relationship between the energy propagating down the tube to that fraction of energy reflected back up the tube. A series of experiments have therefore been conducted using plane waves within an acoustic tube to determine those aspects of a sound wave which cause motion of various types of structures.

III. EXPERIMENTAL PROGRAM

The experimental program was designed to study the motion of several types of structures when subjected to a sound field in which the pressure and pressure gradient could be varied independently. The experimental facility consisted of an acoustic tube as illustrated in Figure 1. The tube has a cross section which is 6 inches square and a length of 25 feet. These dimensions were chosen to allow for acoustical measurements between 100 and 500 cycles per second. Furthermore, samples could be designed having resonant frequencies within this range and having thicknesses up to 0.1 inches. It was desirable to have sufficient sample thickness in order to avoid structures which would be simple resonant membranes.

For most experiments, the sample was inserted in the tube at a distance of approximately 10 feet from the distal end of the tube. The termination was a hard 1 inch thickness of lucite provided with a rubber gasket around its periphery. The behavior of the tube was tested with this termination and it was found that a standing wave ratio of at least 30 decibels was possible for frequencies ranging from 100 to 500 cps. The termination could be moved to any location from the end of the tube up to the sample over a range of 10 feet. For the lowest frequency the termination could be moved at least two half wave lengths. As the termination was moved, the standing wave pattern within the tube could be caused to move and thus vary the pressure and pressure gradient at the location of the sample.

For each sample that was chosen for study, the standing wave pattern within the tube was explored to assure that the presence of the sample did not markedly alter the sound field.

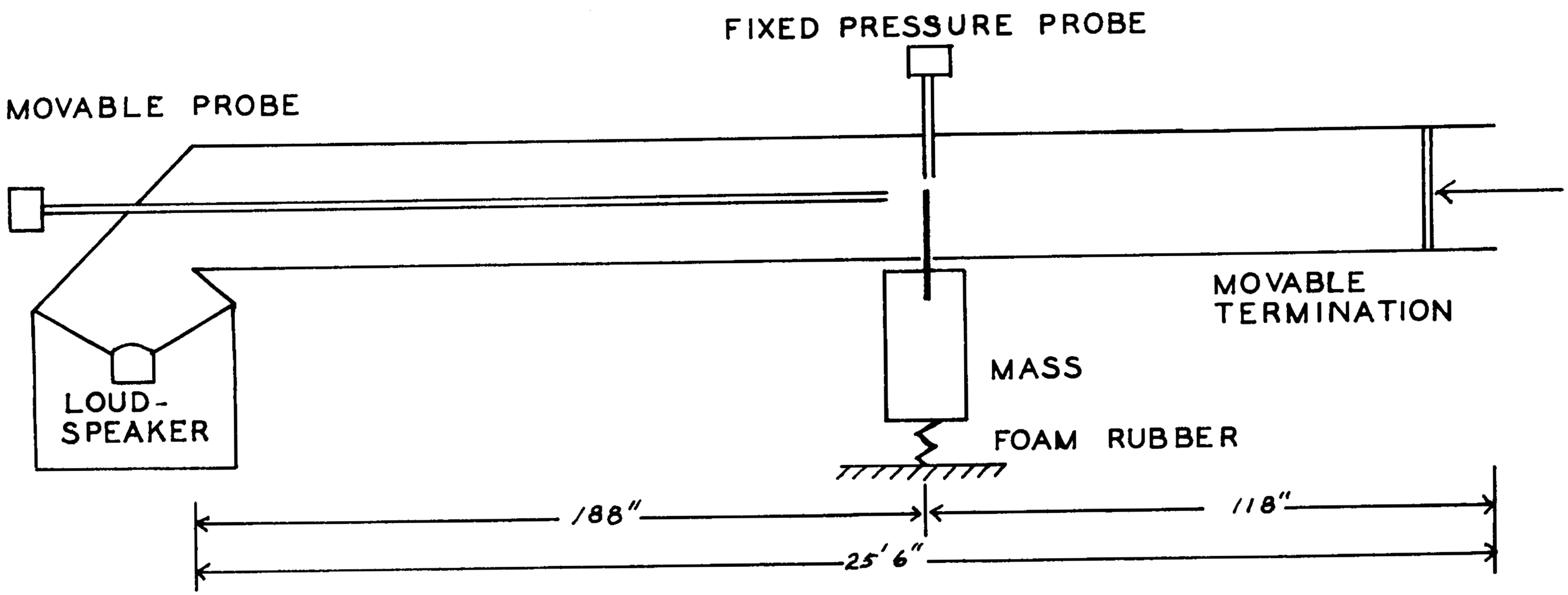

Figure 1. Diagram of Acoustic Tube for the Excitation of Samples.

In general, the sample dimensions were held to 1 by 5 inches with the thickness and mounting being varied to cause different resonant frequencies and modes of vibration. Under these conditions, it was impossible to detect a change in the pressure distribution within the tube due to the presence of the sample. At the beginning of the experimental program, the instrumentation for the acoustic tube consisted of two pressure sensitive probe microphones. The first of the microphones was fixed at the location of the sample to measure the pressure at the sample. The second microphone was introduced at the source end of the tube and could be moved to measure the pressure at any location up to the position of the sample. The second probe microphone could therefore measure the standing wave ratio and the location of the nulls defining the distribution of pressure within the tube. Both microphones were calibrated to allow for an absolute measurement of sound pressure.

The first sample to be studied was that of a simple cantilever beam whose dimensions were 1 by 5 inches and 0.1 inch thickness. The resonant frequency was computed and the actual frequency was measured by shock excitation and an observance of the frequency during the decay. The measured frequency was within 5 per cent of the computed frequency. The sample was then placed within the tube through a port located 10 feet from the end of the tube. Considerable care had to be followed for the isolation of the sample and its reference mass from the tube structure to assure that the excitation of the sample was due to acoustic energy and not due to mechanically borne vibration. A mass, several hundred times the mass of the sample was used as a reference mass. The reference mass was isolated from the main structure by the use of foam rubber.

Measurements were made of the sample vibration, the sound pressure at the sample, the location of pressure nulls, the standing wave ratio and pressure at both the points of maximum and minimum pressure. These measurements were made as a function of the location of the movable termination. When the termination was moved down the tube it was possible to establish various ratios between pressure and pressure gradient at the location of the sample. It was further possible to operate the tube at various power levels so that the net power flow down the tube could be controlled. In addition, it was possible to vary the impedance of the termination so that the ratio between pressure and pressure gradient could be varied.

The first observations consisted of measuring the pressure at the sample and the sample motion for various locations of a hard termination of the tube. The motion of the sample was detected by a frequency modulated capacitance probe for which the output is proportional to the displacement of the sample. As the termination was moved with respect to the sample, the pressure varied in an essentially sinusoidal manner and the displacement of the sample varied in a similar manner, but the maximum sample displacement occurred for a minimum pressure and the minimum displacement occurred for a maximum pressure. The variation was not a smooth predictable function due to adverse loading of the driving loudspeaker which was a function of the length of the tube. Although various means of keeping constant excitation were attempted, the most satisfactory procedure was to maintain a constant maximum pressure in the tube. Under these conditions, the pressure at the location of the sample varied in a predictable fashion versus the location of the hard termination. The motion of the sample varied in a similar predictable manner but out of spatial phase with the pressure. That is, the maximum motion occurred when the pressure was a minimum and the minimum occurred when the pressure was a maximum.

A second problem was encountered in as much as the apparent resonant frequency of the sample appeared to be unstable. A Hewlett Packard oscillator was used to excite the tube and its stability was confirmed by the use of a frequency monitor. It was necessary to measure the frequency to 0.1 cycles per second in order to note the variation in resonant frequency as the termination location was varied. In order to determine the actual frequency of the sample it was necessary to modify the oscillator with a vernier capacitor to allow for this small variation in frequency. The frequency of excitation was varied to obtain a maximum motion of the sample for each location of the termination. It was found that the apparent resonant frequency varied by 0.8 cps out of 212 cycles per second. Data was also taken of the sound pressure at the sample and the sample motion while maintaining constant maximum sound pressure in the tube and adjusting frequency for maximum sample motion. Under these conditions the sample motion was a smooth function of the location of the termination with respect to the sample.

Since it is possible to compute the pressure gradient, the particle displacement and the particle velocity, when the pressure variation within the tube is known, it is not necessary to provide sensors for each of these functions. It is easily shown that the particle velocity and the pressure gradient follow a sinusoidal distribution, with a spatial shift of a quarter of a wave length between these functions and the pressure distribution within a standing wave tube. The motion of a simple cantilever structure, therefore, responds in proportion to either the pressure gradient or the particle velocity within the tube. It was therefore illustrated that the motion of a cantilever sample is in proportion to the pressure gradient when the sample is normal to the direction of wave propagation in a plane wave sound field.

It was interesting also to study the cause of the apparent shift in frequency of the resonant sample, since if this is possible under laboratory conditions, it is also possible under actual conditions. The sample was placed within a chamber that could be evacuated and the sample was caused to vibrate by driving the free end of the sample with an electrostatic probe. The motion of the sample was sensed with the frequency modulated probe, whose output was amplified and fed to the driving probe. The gain was sufficient to cause motion of the sample at its resonant frequency. The frequency of the sample was monitored and measured as a function of the atmospheric pressure surrounding the sample. It was found that the resonant frequency varied over the same range as was noted in the acoustic tube. It is therefore apparent that the air loading on the sample causes a shift in the resonant frequency of the sample. When the adjacent particles are in phase with the sample displacement, the sample acts as if it is in vacuum and the resonant frequency shifts. It is therefore possible to have a structure whose modes of vibration change under certain conditions of excitation.

Similar measurements of the motion of samples vibrating in other modes were performed. The samples included fixed-

fixed samples whose frequency of vibration is equivalent to a free-free bar. This type of sample was chosen since it could be incorporated within a frame for which the back could be left open or it could be enclosed to effectively remove excitation from the back of the structure. Several samples of each type were studied where the thickness was varied so that measurements could be made over the frequency range of 100 to 500 cycles per second. For all samples which were studied, the sample motion was proportional to the pressure gradient or particle velocity at the location of the sample. Since the variation in pressure occurs one quarter wave length displaced from the pressure gradient, the sample motion did not correspond to pressure. It should be understood that the measurements are made at the frequency of maximum motion of the sample. Under these conditions, the sample exhibits a low acoustic impedance, rather than a high acoustic impedance which occurs at frequencies for which the sample exhibits little or no motion. Even though a sample was chosen where the back of the sample was sealed from the sound wave, the motion corresponded to the spatial distribution of pressure gradient rather than the pressure.

The samples which were fixed at both ends introduced new problems in the observation of their behavior within the acoustic tube. The mounting was accomplished by laying the sample on top of an aluminum box structure which has 1/4 inch thick walls. The sample's length was identical to the length of the box and the width was just sufficient so that there was a small clearance between the sample and the sides of the box. Clamping plates were placed above the sample on each end and two screws were inserted through the clamping plates as illustrated in Figure 2. This procedure provides a well defined sample length. No attempt was made to apply tension to the sample during assembly so that it would more closely duplicate a typical structure.

When the sample was placed in the acoustic tube it was found that the frequency of maximum motion was well defined at small amplitudes of excitation but was poorly defined for large amplitudes of motion. For large amplitude excitation it was possible to increase the frequency near resonance and to continue to observe a larger amplitude of motion. Similarly,

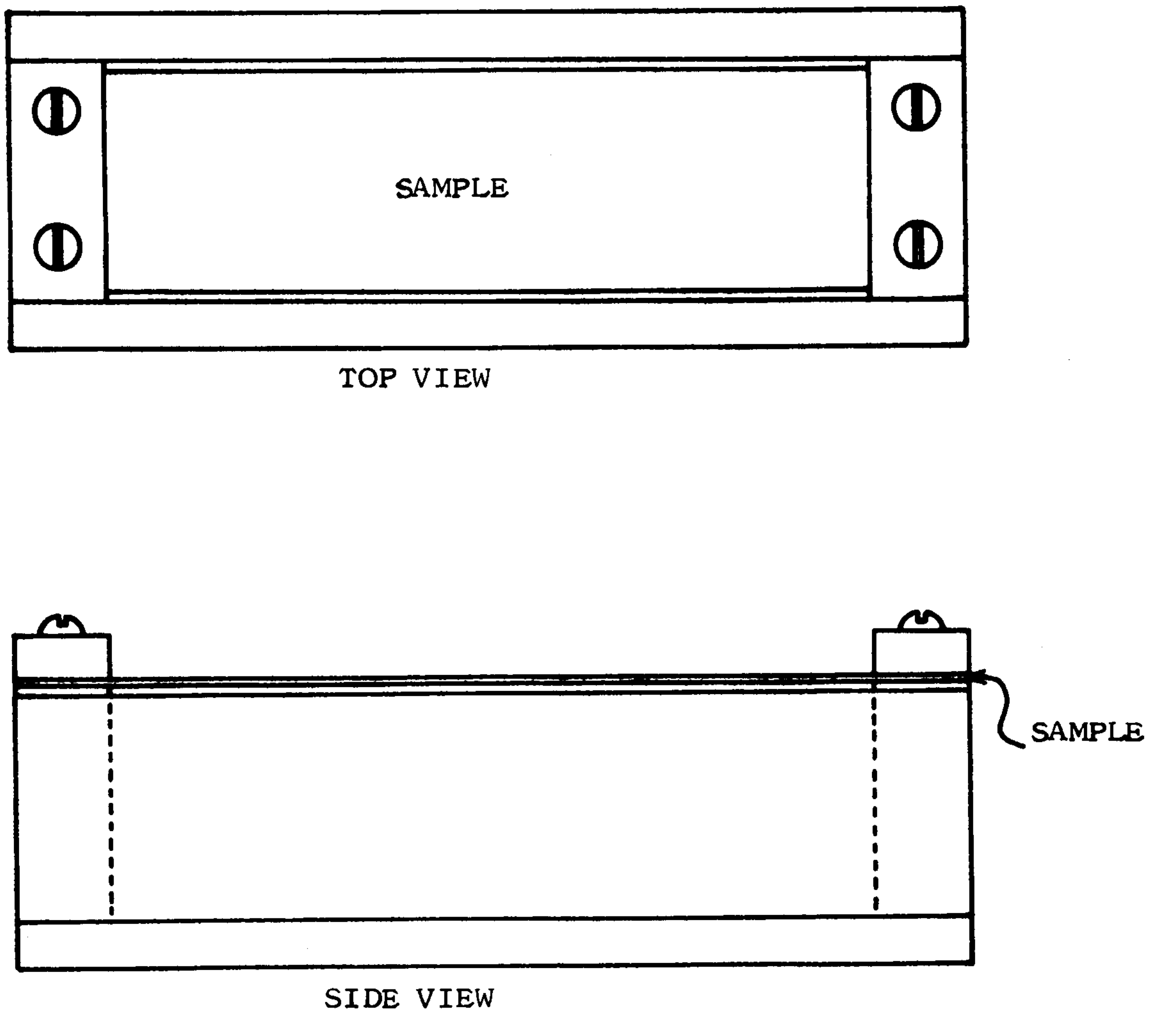

Figure 2. Diagram of Fixed-Fixed Sample with Back Enclosure Illustrating Method of Clamping.

with decreasing frequency the same observations were noted. There was no apparent resonant frequency but a range of frequencies. If a frequency within this range was selected and the location of the termination of the tube was varied, a similar behavior of the sample at constant frequency was observed. These observations are illustrated in Figure 3. It is noted that moving the termination toward the sample results in a different behavior than that obtained when moving the termination away from the sample.

For large amplitude excitation it was possible to observe the motion of the sample versus the location of the termination or versus frequency. The amplitude suddenly increased as indicated in the Figure so that it was obvious that there was sample motion. Similarly, the amplitude of motion would suddenly decrease which was also visible. Supplemental measurements were made on similar samples for which a tension was placed on the sample during the assembly procedure. When sufficient tension was applied a single resonant frequency was obtained. It, therefore, appears that the stiffness of the sample changes as a function of amplitude of vibration. For small amplitudes the stiffness of the material in flexure near the clamps dominates but for large amplitude motion the total stiffness on the structure becomes important. For large amplitude motion it is therefore possible to continue to find a more favorable frequency as the amplitude of motion is either increased or decreased. Since most aircraft exterior structures are similarly constructed, this observation is of importance when observing the acoustic excitation of structures. Similar observations have been reported by Kirchman and Greenspan.[2]

Having established the fact that the motion of various types of samples does not correspond to sound pressure, it was desirable to determine whether the sample motion corresponds

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[2]Kirchman, E. J. and Greenspan, J. E., "Nonlinear Response of Aircraft Panels in Acoustic Noise," The Journal of the Acoustical Society of America, 29, 854 (July 1957).

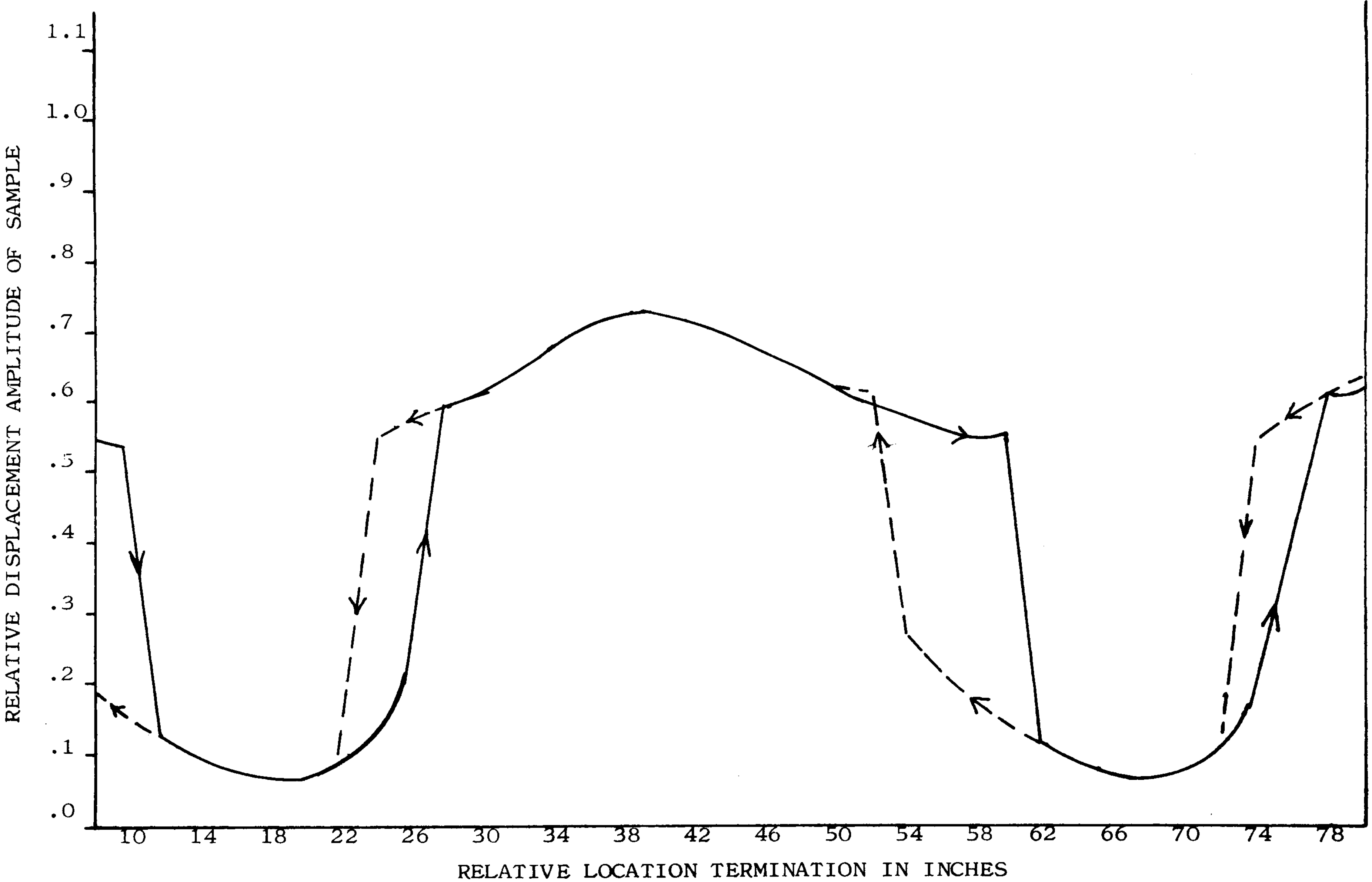


Fig. 3. Response of Fixed-Fixed Sample Showing Different Amplitude Behavior When Termination is Moved Toward versus Away From the Sample.

directly in magnitude to the pressure gradient or particle velocity. Commercially available pressure gradient microphones are limited to use in low intensity sound fields. Typical commercial microphones become nonlinear at sound pressure levels of 120 db in a plane progressive wave. The samples which were selected for study required at least 120 db in order to have a measureable displacement. It was therefore decided to develop a pressure gradient microphone in a manner similar to those pressure microphones which have been developed for high intensity measurements.

A microphone was constructed as illustrated in Figure 4. The sensitive element consists of an aluminum coated mylar diaphragm which is stretched across a brass backplate having a slightly conical section. The interior of the backplate is removed, so that both sides of the diaphragm are exposed to the sound field. For equal pressure on both sides of the diaphragm there should be no motion, whereas when a pressure gradient exists, the diaphragm should exhibit motion. The change in capacitance between the aluminized diaphragm and the insulated backplate causes a change in charge when the microphone is electrostatically polarized.

Several versions of the pressure gradient microphone were constructed until a mechanically stable structure was obtained. The final version of the microphone is illustrated in the Figure. Two types of measurements were performed on the microphone, the first to determine its linearity and the second to determine that the response was that of a pressure gradient rather than a pressure microphone. At several frequencies between 100 and 500 cycles per second the microphone was placed within the acoustic tube and the termination was moved to cause a maximum output of the microphone. This occurred when the pressure at the microphone was a minimum and the maximum pressure occurred one quarter wave length away. A pressure microphone was placed at the pressure maximum, one quarter wave length away, and the output of both the experimental pressure gradient microphone and the pressure microphone were recorded as a function of the excitation of the acoustic tube. The output of the pressure gradient microphone increased linearly with the excitation and with the output of the pressure microphone to the maximum capability of the

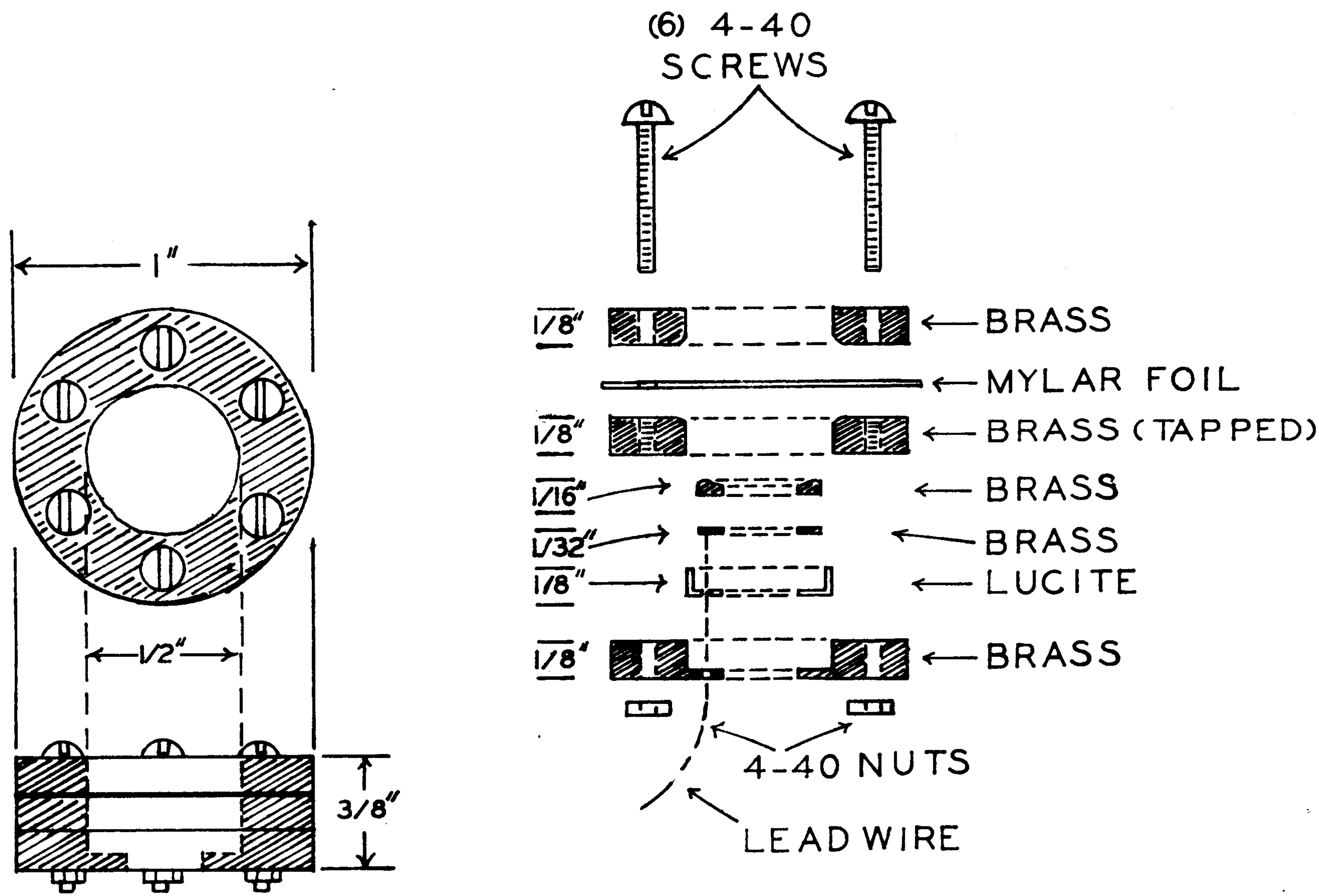


Figure 4. Construction of Velocity Microphone.

driver. This corresponded to pressure gradients equivalent to sound pressure levels in excess of 150 decibels at all frequencies between 100 and 500 cycles per second. The actual particle velocity is computed to be approximately 150 cm per second.

Although it had been clearly established that the output of the pressure gradient microphone corresponded to pressure gradient or particle velocity, additional observations were made by moving the termination and observing the output of the microphone with relation to the standing wave pattern within the tube. These observations clearly indicated that the microphone responds to pressure gradient or particle velocity which is displaced by one quarter wave length from the pressure pattern within the tube.

The next series of measurements was concerned with the absolute magnitude of sample motion under varying conditions within the acoustic tube compared to the motion which occurs when the sample is subjected to a plane progressive wave under free field conditions. Each sample was placed in free space and excited at resonance by a loudspeaker. The motion of the sample was measured and the output of the fixed pressure probe and the pressure gradient microphone were recorded when the two types of microphones were placed in the same orientation with respect to the sample as would be used within the acoustic tube. The sample, the fixed pressure probe microphone and the pressure gradient microphone were then mounted within the acoustic tube.

The termination was set at the end of the tube and the excitation of the tube was adjusted to obtain the same sample vibrations as was obtained external to the tube under free field conditions. A measurement of the pressure at the sample and the pressure gradient were then obtained. The termination was then successively moved to various positions within the tube, resulting in a variation in pressure and pressure gradient at the sample. For each position of the termination, the excitation was readjusted to obtain constant motion of the sample. For all conditions of the standing wave pattern, the pressure gradient at the sample remained constant and at the value observed under free field conditions. The pressure varied over a 45 decibel range as depicted in Figure 5 for a cantilever sample. It can be seen that the sample motion is independent of the sound pressure at the sample. The maximum error compared to free field conditions was 25 decibels. Each of the samples studied previously were subjected to the same measurement procedure with comparable results. In each case the pressure gradient microphone predicted the sample motion under all conditions of the standing wave pattern within the tube relative to the motion observed under free field conditions. Furthermore, the pressure measurement indicated considerable error for each of the samples. It is to be noted that these observations are for samples subjected to plane waves for which the sample is small enough that the disturbance of the sound field is undetectable.

Within the small cross section of the tube, it is difficult to design samples whose geometry is comparable to that used in actual structures without disturbing the sound field. Although simple samples having various modes of vibration are easily studied, it was felt that a more typical structural section should be studied before reaching definite conclusions.

An additional series of measurements was performed on a series of samples which fully obstructed the tube. For the first exploratory measurements, samples were constructed of sheet aluminum having various thicknesses and mounted in a framework which fully closed the 6 by 6 inch cross section of the tube. The samples were placed at the termination of the tube with the back side exposed to free space.

Under these conditions, the resonant frequency of the sample was determined by observing the motion of the center of the sample as the frequency was varied. The acoustic impedance of the sample was determined by measurements of the standing wave pattern, both location of the nulls and the standing wave ratio, for frequencies near and at the resonance of the sample. These measurements indicated a sharp shift in impedance at frequencies near the resonance of the sample. For frequencies at which the sample exhibited little or no motion, the pressure at the sample doubled and the impedance was equivalent to a hard termination. It was impossible to interpret directly the results of the measurements due to the fact that the edges of the sample were fixed under all conditions. When the sample

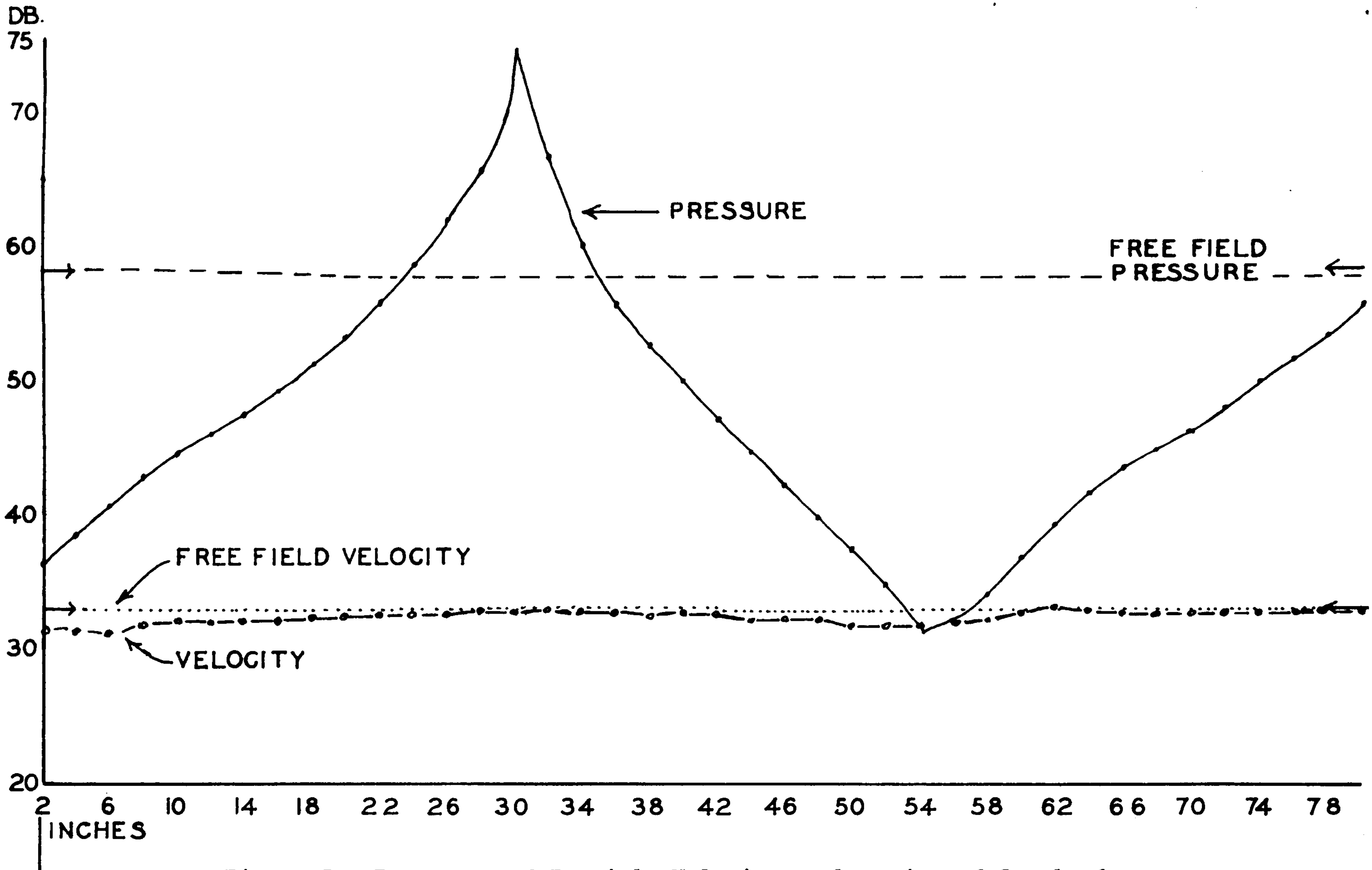


Figure 5. Pressure and Particle Velocity at Location of Sample for Constant Sample Displacement. Free Field Pressure and Particle Velocity are Indicated for the Same Sample Displacement.

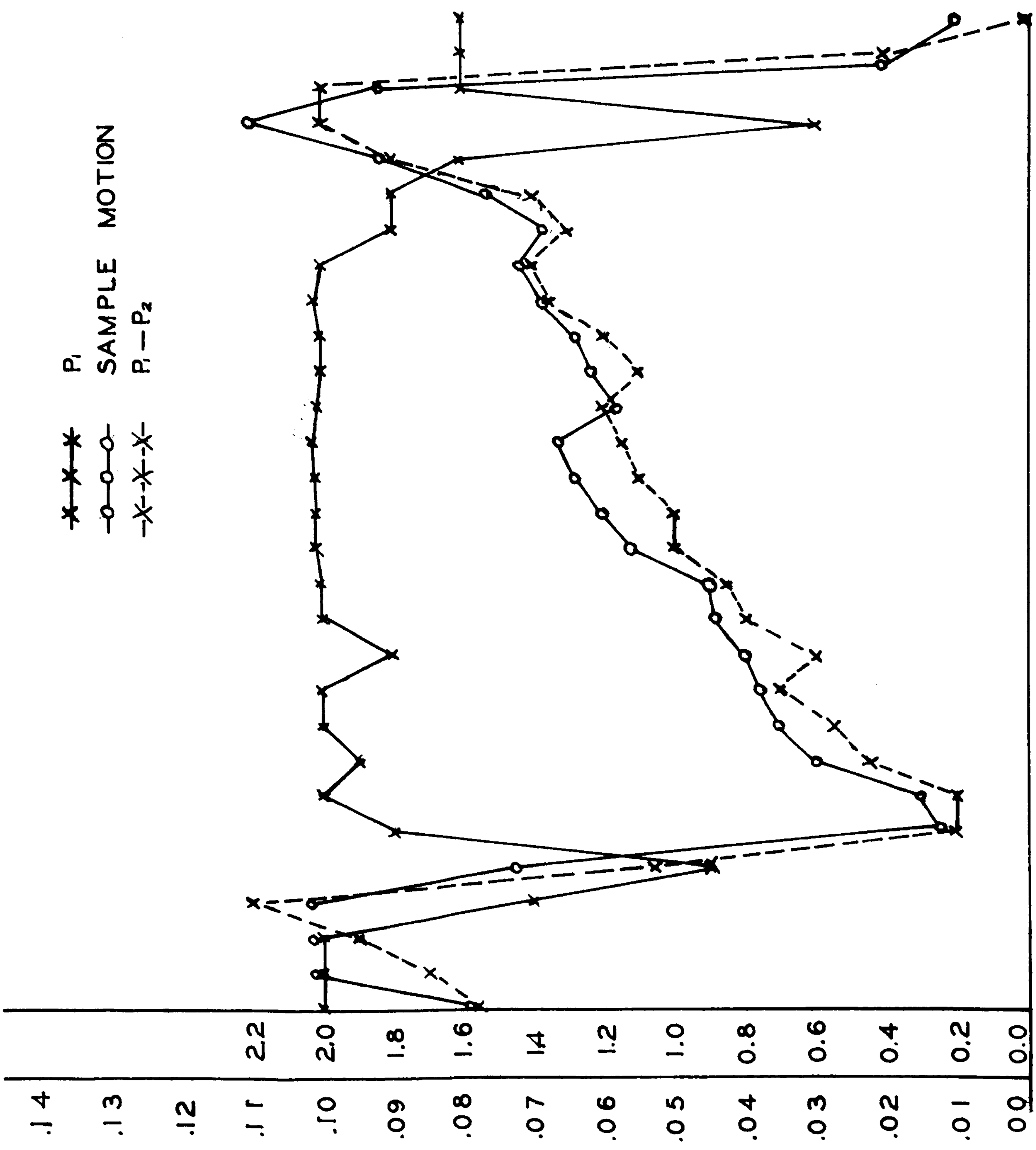
was vibrating, the impedance of the central portion of the sample was extremely low and the edges was extremely high. The result was that the wave front directly in front of the sample was no longer plane but had considerable curvature as observed with a probe microphone. It is clearly apparent that the impedance of the portion of the sample which is free to vibrate is extremely low and will not support a pressure.

In order to investigate further this type of sample, the sample was placed ten feet from the end of the tube at the same location as the smaller samples. The hard termination was placed at the end of the tube and the excitation was adjusted for maximum sample motion. As the termination was moved relative to sample, the motion of the sample was caused to vary. Under these conditions, that is at sample resonance, considerable energy is transmitted through the sample to the tube beyond, providing that the acoustical impedance at the sample as determined by the termination of the acoustic tube is low. This condition occurs when the hard termination is at a distance from the sample corresponding to odd multiples of a quarter of a wave length.

In order to observe the cause of the sample motion under these conditions, two pressure probe microphones were introduced, one in front of and one directly behind the sample. The magnitude of the pressure in front, the magnitude of the pressure behind, the motion of the sample, and the phasor sum of the pressure in front of and behind the sample were recorded as a function of the distance between the sample and the hard termination. The results of these measurements for an aluminum sample whose thickness was 0.02 inch are given in Figure 6. It is first noted that the phasor sum of the pressure on the two sides of the sample varies in proportion to the sample motion. The pressure immediately in front of the sample, on the source side, does not vary in propotion to the sample motion. Furthermore, it is seen that under certain conditions, the pressure behind the sample dominates in causing the sample motion.

For samples vibrating in their fundamental mode, that is, where all parts move in the same direction at the same instant, the motion is dependent upon the instantaneous pressure

Figure 6. The Incident Pressure, P_1, the Difference in Pressure Across the Sample and the Sample Displacement as a Function of Distance to the Termination of the Tube.

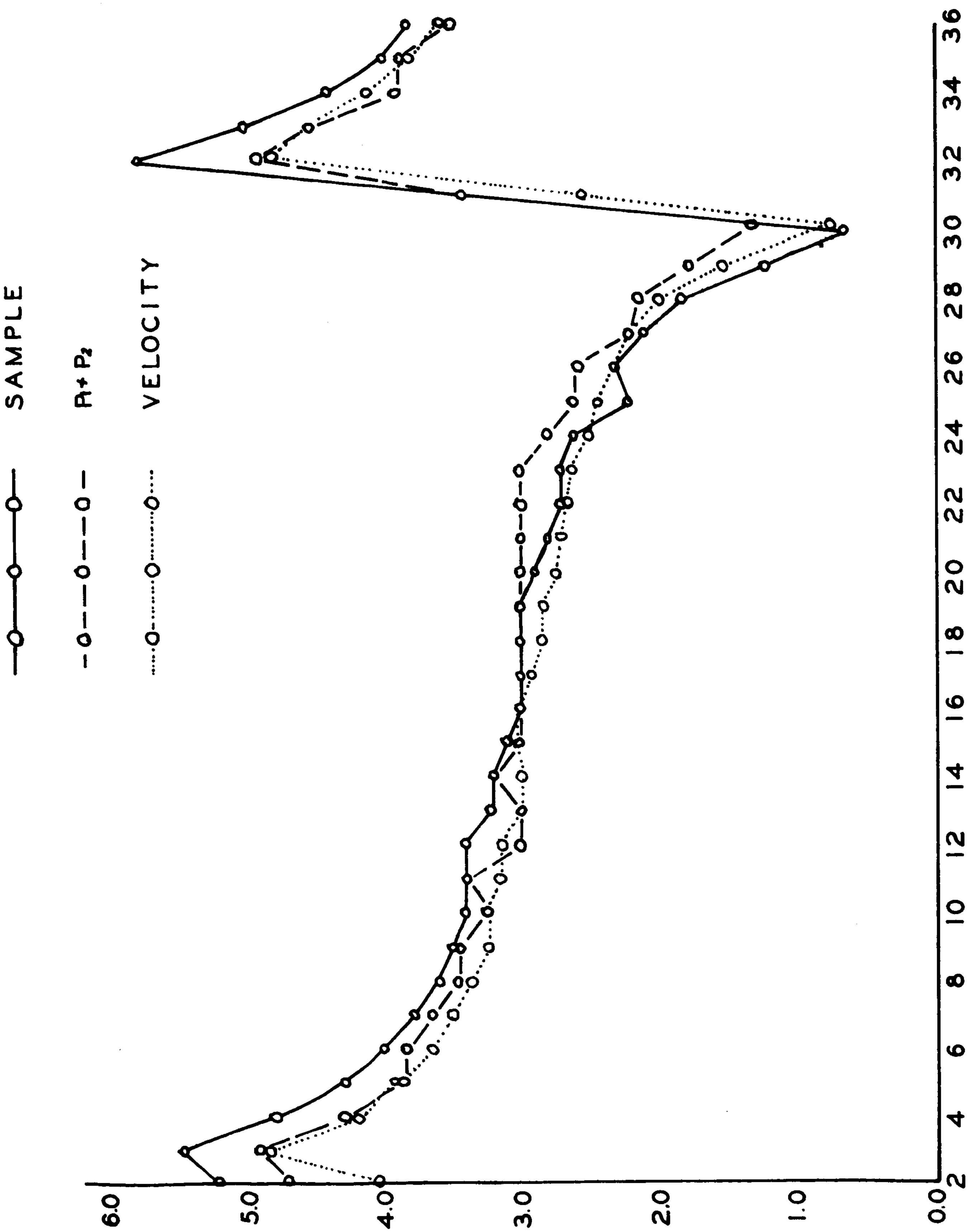


differential across the sample and independent of the pressure directly in front of the sample. Under these conditions the sound field is altered by the sample and the location of the termination behind the sample. It is therefore impossible to measure the sound field before the sample is inserted to determine the response that will be measured.

Even though the sample alters the sound field within the acoustic tube, it is of interest to measure the pressure gradient immediately in front of and on the source side of the sample. The pressure gradient microphone which was constructed was placed near the center of the sample and the output was measured along with sample motion for various positions of the termination. It should be understood that the pressure gradient microphone may also be considered to be a particle velocity microphone and it is not surprising that the particle velocity immediately in front of the sample corresponds to sample motion. A graph showing the particle velocity and the sample motion for various locations of the termination are shown in Figure 7. It should again be noted that these measurements are performed at the frequency for which the sample exhibits a maximum motion and the acoustical impedance of the central portion of the sample is extremely low. For this experiment it is further noted that the waves immediately in front of the sample are no longer plane and are distorted due to the fact that the edges of the sample are fixed and the center of the sample is free to vibrate. It is further noted that if the sample is excited at a frequency for which little motion is observed then the pressure doubles at the sample corresponding to a hard termination and the particle velocity is extremely low. Even under this condition the sample motion corresponds to particle velocity rather than the pressure.

Since all of the measurements performed thus far indicate that the various types of samples respond to the pressure gradient it was decided to perform some measurements under free field conditions which might further substantiate these findings. It is known that the pressure gradient and particle velocity exhibit directional properties for which their magnitude is a maximum in the direction of propagation of the sound wave. Furthermore, a minimum should exist perpendicular to the direction of propagation. The same rectangular samples clamped around their edges were subjected to a free sound field created

Figure 7. Particle Velocity Immediately in Front of a 6" x 6" Square Sample. Sample Motion and Differential Pressure Across the Sample are in Agreement with Particle Velocity.



by a loud speaker. The pressure gradient microphones were located at the position of the sample and the output of both microphones and the motion of the sample were recorded versus the orientation of the sample with respect to the direction of propagation of the sound wave. For constant excitation of the loud speaker the sound pressure at the location of the sample remains constant. The output of the pressure gradient microphone is typical of a bi-directional microphone, that is, a cosine function, having a null when the microphone is oriented at 90^{o} with respect to the direction of propagation. Similarly, the motion of the sample follows a bi-directional pattern with a minimum motion occurring when the sample is oriented 90^{o} to the direction of propagation. For this particular experiment the resonant frequency corresponded to a wave length which was much larger than the dimensions of the sample. It is, of course, possible to produce various types of patterns depending upon the relationship between wave length and sample size and also dependent upon the particular mode of vibration of the sample. This type of experiment further substantiates the fact that it is necessary to define the directional properties of the sound field in order to predict the motion of a mechanical structure.

During the experimental program no attempt was made to produce fatigue failure of samples. It has been assumed that for a sample to fail by fatigue it is necessary to produce motion of the sample. All of the experiments have therefore been conducted on a basis of relating the motion of the sample to the sound field rather than an actual study of acoustic fatigue. A second restriction on the experiments is concerned with the use of single frequencies for which each sample exhibits motion. Under actual conditions the source is most likely to be a random noise rather than a single frequency. The use of random noise would complicate the interpretation of the results which have been obtained for this particular study.

All of the experiments which have been performed during this program have resulted in a correlation betwen either pressure gradient or particle velocity and the motion of the sample when the excitation is restricted to plane sound waves at single frequencies. The absolute magnitude of the sample motion under varying conditions of standing wave patterns correspond to the sample motion which is produced under free field conditions. A measurement of sound pressure rather than pressure gradient results in large discrepancies between the free field sound pressure and that obtained within the standing wave tube for constant sample motion. These discrepancies are presumably due to the fact that the sample has an equivalent acoustical impedance which is low at frequencies for which the sample exhibits motion. For frequencies at which samples exhibit small motion the particle velocity immediately in front of the sample still corresponds to sample motion. Although it is impossible to separate pressure gradient and particle velocity, it is assumed that the forcing function for a sample is the instantaneous pressure differential across a sample. In cases where it is inconvenient to measure the pressure differential, the particle velocity immediately adjacent to the sample corresponds to sample motion. Further experiments under free field conditions indicate that the sample motion is dependent upon the orientation of the sample with respect to the direction of propagation of the sound wave. It is therefore the conclusion of these experiments that it is necessary to measure the directional aspects of a sound field in order to determine whether or not the sound field simulates another sound field for the purposes of studying acoustic fatigue. As with pressure measurements it may be necessary in certain types of simulation facilities to measure the pressure gradient or particle velocity with the structure present since it is likely that there may be some interaction between the structure and the sound field.

IV. RANDOM NOISE EXCITATION

During the experimental program it was possible to observe the resonant characteristics of various types of structures. In general, the measurements were performed at the frequency for which maximum sample motion existed. In order to tune the oscillator to the exact frequency of a given mode of vibration, it was necessary to modify a commercial oscillator with a vernier capacitor in order to allow for setting the oscillator at the sample frequency. This is indicative of the high Q of the samples showing that the bandwidth is extremely narrow. It is therefore anticipated that the motion of a sample is highly dependent upon the time history of the excitation. For plane wave random noise excitation the sample's motion will correspond closely to that of a highly selective filtering system. For a sound field in which the incidence of the sound energy varies with time, a further effect on the sample motion should be noted since it is the component of the sound field which is normal to the sample which provides the excitation.

It should be noted that the directional characteristics of the sound field are selective in the sense that the motion of a sample is a function of the orientation of the sample with respect to the sound field. The further selectivity of the sample due to its high mechanical Q indicates that the behavior of the sample for random excitation will be a function of the randomness of the direction of propagation as well as the time history of the amplitude variation. In assessing a sound field, consideration must be given to the nature of the sound source with respect to directional characteristics as well as the frequency distribution of the sound energy.

V. SIMULATION FACILITIES

The second objective of the present program was to assess simulation facilities with respect to the findings of the experimental program. The literature describes a number of facilities which are currently in use for acoustic fatigue studies. Further information is given describing the actual sound fields which are found for both aircraft and missiles. In order to assess the simulation facilities, it is necessary to compare the actual sound field to that produced within the simulation facilities. For this reason the actual sound field will be discussed first.

A review of the literature shows that numerous studies have been made of the noise field surrounding both jet and rocket engines. The objective of these studies have ranged from the problem of annoyance of people to the actual mechanical fatigue of the aircraft structure. During 1952 a Symposium[3] held at

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[3]Aircraft Noise Symposium, San Diego Meeting of the Acoustical Society of America, November, 1952.

the acoustical Society meeting in San Diego described the aircraft noise problem. This Symposium on aircraft noise defined the problem as the effect of noise upon people. No mention of adverse effects of structures was noted. The papers presented at this Symposium and other papers appearing at this time described the noise field about various jet engines which are of use in the present problem.

The observations of von Gierke[4] were concerned with the total acoustic power radiated by a jet noise source. The reported measurements were concerned with radiation patterns as well as the frequency distribution of noise. Hubbard and Lassiter[5] presented experimental information concerning the location of the apparent noise source for a jet engine. Other papers were concerned with the control of these noise sources and the isolation of the sources from personnel.

Additional papers presented by both the Air Force and Navy give considerable information concerning the sound field surrounding a jet engine. For example, Kennard[6] presents information of the overall sound pressure level in the near vicinity of a twin engined jet aircraft. The information presented indicates that the dominant sound source exists at a location several diameters behind the jet engine. The contours of constant sound pressure level indicate that the sound source is complex and the direction of wave propagation in the vicinity of the actual aircraft structure varies markedly along the

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[4]von Gierke, H. E., "Physical Characteristics of Aircraft Noise Sources," The Journal of the Acoustical Society of America, 25, 367 (May 1953).

[5]Hubbard, H. H. and Lassiter, L. W., "Experimental Studies of Jet Noise," The Journal of the Acoustical Society of America, 25, 381 (May 1953).

[6]Kennard, D. C., Jr., "Sonic Vibration as Exemplified by the RB-66B Airplane," The Journal of the Acoustical Society of America, 30, 462 (May 1958).

structure. In this particular paper, relationships are established between vibration levels and sound pressure levels for various parts of the aircraft. The measurements are presented as a function of frequency indicating that the maximum vibration occurs for frequencies ranging from 300 to 600 cycles per second. Although no direct measurement is given in this paper, it is apparent that the maximum vibration also occurs at locations where the sound energy strikes the structure at normal incidence.

Additional measurements performed under sponsorship of the Navy indicate similar sound pressure levels and distribution around aircraft as reported by Pietrasanta.[7] These measurements show the variation in radiation pattern surrounding a jet engine as a function of frequency. In commenting on the problem of noise measurement around a jet aircraft, Pietrasanta indicates that the noise field can be predicted at distances from 75 to 200 feet from the aircraft, closer than 75 feet one encounters the "near field" of the engine. Since a large portion of the aircraft exists within the near field of the noise source, it is these measurements that are of most importance for acoustic fatigue.

Similar types of measurements have been made for rocket engines. Mayes has reported measurements for rocket engines in the near field which would be of importance for acoustic fatigue studies. His measurements indicate that the directivity patterns vary markedly in the near vicinity of the sound sources and that the apparent source exists at a considerable distance behind the location of the rocket nozzle. It would appear that extensive measurements have been made for both jet and rocket engines which should provide sufficient information to allow for the evaluation of the actual sound field. All of the measurements which have been reported are based upon the measurement of sound pressure level as a function of frequency and as a function of orientation with respect to the noise source. In general, the measurement is made for a frequency bandwidth of one octave with the exception of some discreet frequency measurements of components of the noise which

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[7]Pietrasanta, A. C., "Noise Measurements Around Some Jet Aircrafts," The Journal of the Acoustical Society of America, 28, 434 (May 1956).

may be created by a compressor. A summary of the observations that are made can be referenced to early measurements by von Gierke. These early measurements were made in the near vicinity of jet engines and accomplished by measuring the sound pressure level in octave bands and for various orientations around the jet source. The two assumptions that were made were that (1) the directivity patterns are rotationally symmetrical and (2) the pressure measurements are made in the far field which means that pressure and particle velocity are nearly in phase. At the distances of concern, that is within the near field of the engine, the second assumption is not justified. The observations which are reported indicate that the directivity of the sound source varies within the range of interest and that the pressure and particle velocity are therefore not in phase.

The nature of simulation facilities has been influenced greatly by the commercial companies which are capable of supplying large amounts of acoustical power. The types of facilities which are currently in use are of various types as follows: (1) Actual full scale sound source consisting of a jet engine instrumented for static testing or a turbulent noise source as produced with a blow down air source and appropriate nozzles or its use of a high powered siren. (2) A plane wave tube which is driven by multiple loud speaker driver units and is terminated in its characteristic impedance. (3) A standing wave tube which is used primarily for the determination of microphone linearity. (4) A reverberation room similar to those used for evaluating acoustical materials but incorporating high powered loud speaker elements. A variation is to use a siren or other noise source for excitation of the room. (5) A reverberant box which is essentially a hard walled chamber similar to a reverberation room but the dimensions are small compared to a wave length over most of the frequency range. (6) A low frequency chamber in which the input is provided by a large diameter hydraulic actuated piston.

Before discussing each of the above types of facilities it is desirable to discuss the assumptions which are used for the various types of facilities described above. For structures which exist on the exterior part of the aircraft it is normally assumed that the sound field is a free progressive field with either a spherical or plane wave front. For structures which are to be located internal to an aircraft or missile, it is assumed that the sound field approaches that of a reverberant sound field where the energy flow is random in direction as well as amplitude. For exterior sound fields it should be remembered that the literature reveals that most of the aircraft or missile structure is in the near field of the sound source and that the pressure and particle velocity are not necessarily in phase. The directivity patterns and the decrease in pressure versus distance indicates that this assumption is invalid over most of the structure. The literature further indicates that the actual source of sound energy exists at a considerable distance behind the exit nozzle so that the direction of propagation and thus the angle of incidence of the energy for a given part of a structure is unknown.

For the internal sound field the measurements reported are usually octave band measurements of the pressure throughout a given compartment of the aircraft or missile. In general, it is found that the sound pressure distribution is relatively uniform throughout these compartments and it is therefore concluded that the sound field is reverberant. It should be pointed out that a similar observation would be made if the sound field of a plane progressive wave is explored at relatively large distances from the sound source. The walls of the structure allow the sound energy to enter a compartment due to their low transmission loss and it is likely that energy flows out of, as well as into, the compartment through the walls. It is to be understood that the energy transmitted through the walls is frequency selective due to the fact that the walls vibrate in specific modes so that their transmission loss is low at selected frequencies and high at other frequencies. Information relating to these specific properties can be obtained from architectural acoustic studies[8] where such structures have been studied from the viewpoint of their use in buildings. In the present study of the literature there

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[8]Watters, B. G., "Transmission Loss of Some Masonry Walls," The Journal of the Acoustical Society of America, 31, 898 (July 1959).

appears to be no supporting evidence that indicates that the sound field within a compartment is reverberant other than the indication of uniformity of the sound pressure field. The dimensions of most compartments are small compared to the wave length of sound which is felt to be detrimental in the design of a space which should result in a reverberant field.

In order to discuss the various types of simulation facilities listed above the assumption will be made that exterior structures are exposed to progressive wave sound fields and that the interior of the structures are exposed to a diffuse reverberant field. During this discussion it should be remembered that the evidence presentedin the literature does not confirm these assumptions. However, it is apparent that the users of simulation facilities have made similar assumptions in the selection of their facilities.

The first type of simulation facility listed above concerns the use of an actual jet engine noise source or an equivalent. For this particular type of simulation facility there is no question that the energy density and the flow of energy do not duplicate that encountered under actual conditions. The use of these facilities involves the determination of the response of various types of panels or structures for various orientations to the flow of acoustic energy. In general, this type of facility is used for destructive tests where actual acoustic fatigue takes place. There is no reason to believe that additional types of acoustic measurements are required to assure that this type of facility duplicates the actual sound field.

The second type of simulation facility is that of the plane progressive wave tube. A description of such a system is given by Hilliard.[9] A long tube of restricted cross section

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[9]Hilliard, John K. and Fiala, Walter T., "Methods of Generating High-Intensity Sound with Loudspeakers for Environmental Testing of Electronic Components Subjected to Jet and Missile Engine Noise," J.A.S.A., 30, 533 (June 1958).

is used to confine the flow of acoustic energy so that the energy density is comparable to that in a free progressive sound field. By restricting the cross section of the tube the power requirement can be limited to that available by electro-acoustical driving units. Computations by Hilliard indicate that a plane wave tube with a two foot diameter and properly terminated can produce energy densities comparable to that encountered in actual practice with an electrical power input of 180 watts. The limitation of the plane wave tube is that the structure or device to be tested must be limited to a size which is sufficiently small compared to the cross section of the tube to minimize a disturbance of the sound field. It is possible within such a facility to vary the orientation of the device under test so that the response for various angles of incidence can be studied. For extremely small devices or structures the plane wave tube is a valuable tool. When the device or structure becomes comparable in size to the dimensions of the cross section of the tube it is necessary to make additional measurements in the tube to determine whether or not the device disturbs the sound field. Measurements can be made with pressure sensitive microphones if the tube is excited at single frequencies and the standing wave pattern is determined with the device present in the facility. For random noise excitation similar measurements can be made at the location of the device with the use of a pressure gradient microphone as described in this report in conjunction with measurements made with a pressure sensitive microphone. The plane wave tube is therefore a useful facility but it is recommended that additional measurements be made when devices or structures tested within the facility become comparable in size to the cross sectional dimensions of the tube.

The standing wave tube such as has been used in the present program has been used primarily for the determination of the linearity of pressure microphones. In selected instances where extremely high sound pressures are desired, the standing wave tube has been used for studies involving the response of structures. From the results of the present program it is evident that the excitation of structures can be accomplished within a standing wave tube. However, the maximum response is obtained at the location of minimum rather than maximum sound pressure. Such a facility can be useful in exploring a particular design when single frequencies are used. It is necessary that measurements of pressure gradient or particle velocity should be made at the location of the structure to determine that aspect of the sound field which is related to motion.

The reverberation room has been used extensively for acoustical testing since 1915.[10] The use of a hard walled room for observing the absorption of acoustical energy has been valuable for the determination of characteristics of acoustical materials. With relatively small amounts of acoustical power input, high sound pressure levels and corresponding energy densities are created due to the fact that the sound undergoes multiple reflection within the room. An example of the levels obtained with relatively low electrical power input is given by McAuliffe.[11] Since numerous reverberation rooms were available throughout the country for the purposes of determining the absorption properties of acoustical materials, it was possible to determine the suitability of these rooms for high intensity noise testing. The majority of articles relating to the design of reverberation rooms for high intensity noise testing are concerned with the determination of the uniformity of the sound pressure level as a function of frequency. Numerous papers have been written concerning similar problems when the objective is to determine acoustical absorption. For example, Waterhouse[12] has written comprehensive

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[10]Benson, Robert W., "Efficiency and Power Rating of Loudspeakers," Proceedings of the National Electronics Conference, Hotel Sherman, Chicago, Illinois, October 3,4,5, 1955.

[11]McAuliffe, D. R., "Design and Performance of a New Reverberation Room at Armour Research Foundation, Chicago, Ill.," J.A.S.A. 29, 1270-1273 (December 1957).

[12]Waterhouse, Richard V., "Output of a Sound Source in a Reverberation Chamber and Other Reflecting Environments," J.A.S.A., 30, 4 (November 1964).

papers concerning interference patterns in reverberant rooms. By an examination of the sound pressure field it can be determined that certain interference effects take place at the boundaries which cause an uneven distribution in sould pressure level. Throughout the literature the reverberation room is described as a structure in which a uniform and diffuse sound field is created. For example, Olynuk and Northwood[13] indicate that the "key hypothesis in reverberation theory is that the sound field is diffuse and the room surface is therefore exposed to an assembly of waves from all angles of incidence." The key assumption is that the sound field is diffuse. This author, as well as others, attempts to describe the conditions which result in a diffuse sound field. In the present review of the literature it has been impossible to find a direct measurement of diffusion within a reverberation room. Some of the requirements are that the room's dimensions must be large compared to a wave length and that the surfaces of the room should be non-parallel. As an example of an attempt to measure diffusion, Balachandran[14] uses the following criteria: (1) An observation of the fluctuations in decay curves as obtained in an empty room; (2) the variation in correlation coefficient with distance between two points in the room, and (3) the absorption coefficient of a standard sample of material for all room conditions and frequencies. Of the three criteria, the determination of correlation coefficients results in the most definitive measurement of the sound field.

The use of correlation methods for the study of sound fields is effective when the sound field is either that of a progressive wave or a diffuse field. Two points in space are

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[13]Olynyk, D. and Northwood, T. D., "Comparison of Reverberation-Room and Impedance Tube Abosrption Measurements," J.A.S.A., 36, 2171 (November 1964).

[14]Balachandran, C. G., "Random Sound Field in Reverberation Chambers," The Journal of the Acoustical Society of America, 31, 1319 (October 1959).

selected for study for which the pressure variation versus time is studied for various delay times introduced between the two signals. For a progressive wave, there is a delay time between the two signals which results in perfect correlation. For a diffuse field, there is no delay time which will establish a correlation for the two signals. For sound fields which contain numerous sources but are neither diffuse nor simple progressive waves, the correlation methods indicate that some correlation exists, but does not indicate the nature of the sound field. These methods therefore have limited application for the study of simulation facilities.

Currently, there is an American Standard[15] being written describing the use of reverberant rooms for high intensity noise testing. This Standard describes current practices concerning the shape and size of the reverberation room. The present draft of the Standard indicates that a structure may be tested within the room providing that the dimensions of the structure are no more than one-half of the dimensions of the room. Under such conditions it may be necessary to make measurements of the sound field with the structure or a mechanical equivalent of the structure in place. The required measurements are made with a pressure sensitive microphone and with a filter having a one-third octave bandwidth. At the present time, instrumentation is not available which would further define the diffuse nature of the sound field. The effect of the structure on the sound field is likely to be obscurred by the fact that the modes of vibration have bandwidths which are small compared to a one-third octave bandwidth and therefore, the interaction between the structure and the sound field is unlikely to be detected.

The fifth environment listed above is a compromise of the reverberation room. A commercial unit is available described as a reverberation box. The reverberation box described by

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[15]Fourth Draft of the Information Document, "High Intensity Acoustic Testing of Equipment." Writing Group S 2/1-W-43. November, 1967.

Hilliard[16] is a small chamber with a hard surfaced interior containing a number of loudspeaker driving units. The term "reverberation" is used to describe the reflective nature of the walls but is not to be confused with the reverberation room where the objective is to provide a diffuse sound field. Throughout most of the audio frequency range the dimensions of the reverberant box are small compared to the wave length of sound. A typical box available commercially has one dimension which is only 18". A measurement of sound pressure within the box indicates that high intensities can be achieved, however, the impedance associated with the pressure is not defined. Throughout most of the frequency range it must be assumed that the pressure gradient is extremely small since the boundary conditions require that the particle velocity vanish normal to wall surfaces. The particle velocity and pressure gradient is expected to be 30 to 40 decibels below the pressure gradient which would be found in either a progressive wave field or in a diffuse sound field. For the boxes which are available commercially it is expected that this condition will exist for frequencies below 1000 cycles per second. Independent of any measurements that might be made within such an enclosure, it is expected that such a facility is of little value for acoustic fatigue simulation. This type of unit has been used extensively by various manufacturers for meeting the requirements of environmental specifications since the unit is available commercially as a package unit.

The greatest difficulty in simulation of the actual noise field of a jet or rocket engine comes for the lower frequency range. The reverberation room is limited due to its finite dimensions. For this reason a special low frequency chamber was constructed at NASA Langley for the purposes of providing a simulation of the sound field for the lower frequency range. The dimensions of the enclosure are such that the wave length of excitation is either greater than, or comparable to, the dimensions of the enclosure. The boundary conditions require

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[16] Hilliard, J. K. and Fiala, W. T., "High Intensity Sound Reverberation Chamber and Loudspeaker Noise Generator," J.A.S.A., 31, 269 (March 1959).

that the particle velocity normal to the surface of the enclosure should vanish. The measurements of the performance of this facility are again made with pressure sensitive microphones for which it is demonstrated that the sound field is uniform and the pressure levels obtained are comparable to that under actual conditions. The pressure gradient and particle velocity within the chamber are again of the order of 30 to 40 decibels less than would be encountered in either a progressive wave field or a diffuse sound field. The response of structures placed within the chamber is therefore expected to be of the order of one one hundreth of the amplitude that would occur under actual conditions. A measurement of the pressure gradient or particle velocity within this facility would reveal the actual expected response of various structures. A modification of the facility would be possible that would result in a large increase in the pressure gradient. If, for example, a second volume is coupled to the main volume of the facility through a large size duct and the two volumes are made to resonate with the mass of air in the connecting duct it would be possible to create large amplitude particle velocities within the duct. A pressure gradient would exist relative to the two chambers due to the alternate storage of energy in each of the chambers. It would, of course, be necessary to provide instrumentation which would be sensitive to the pressure gradient in order to correlate results with the sound field.

Each of the several types of facilities that are currently being used for simulation purposes have been discussed relative to the findings during this program which indicate that the forcing function for a mechanical structure is the differential sound pressure across the structure created by the sound field. A major problem is encountered in determining the suitability of these facilities for the simulation of high intensity noise fields due to the fact that adequate measurements of the actual sound field have not been made. Furthermore, the measurements of the actual sound field leaves some question as to the exact type of sound field that is necessary for various parts of the structure. With these restrictions it is possible to indicate that the reverberant box is inadequate on the basis of design since it provides neither a progressive wave field nor a diffuse sound field. The low frequency facility at Langley in its present configuration should produce misleading results. Modifications to this facility appear possible where the configuration is changed to provide an alternate storage of energy in two volumes. Such a configuration would create pressure gradients which are comparable to those encountered in actual practice.

The reverberation room is by far the most extensively used facility. The results obtained in such a room have been of much value with the major problem concerned with the definition of limitations for the lower frequencies. Gross errors in producing a diffuse sound field are easily recognized but the extent of the transition range between non-diffuse and diffuse is difficult, if not impossible, with present instrumentation.

VI. RECOMMENDATIONS

The objective of the present program has been to determine those aspects of a sound field which produce motion of a structure and thus contribute to sonic fatigue. All of the measurements indicate that the forcing function for a structure is the differential pressure. At the resonance frequencies of a structure, the equivalent acoustic impedance is low and the transfer of energy to the structure is not predicted from a measurement of the pressure field. Since several types of simulation facilities duplicate the pressure field without a corresponding energy density, the behavior of the structure placed within the facility varies from that which would occur under actual field conditions.

The exact nature of the sound field and the degree to which a simulation facility duplicates the actual sound field are difficult to determine with present instrumentation. Although correlation techniques are useful for specific types of sound fields, the actual sound field cannot be described in a manner which is of use in assessing simulation facilities. It is necessary that instruments be developed which allow for a definition of the sound field as a vector quantity, for which the magnitude and direction of the energy flow is obtained.

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